Journal of Engineering Science and Technology
Vol. 6, No. 2 (2011) 146 - 160

HUNTING PHENOMENON STUDY OF RAILWAY CONVENTIONAL TRUCK ON TANGENT TRACKS DUE TO CHANGE IN RAIL WHEEL GEOMETRY

KARIM H. ALI ABOOD*, R. A. KHAN

Department of Mechanical Engineering, College of Engineering & Technology
Jamia Millia Islamia New Delhi-110025, India
*Corresponding Author: karimhali@yahoo.com

Abstract

A mathematical dynamic model of railway conventional truck is presented with 12 degrees of freedom equations of motion. The presented dynamic system consists of conventional truck attached with two single wheelsets in which equipped with lateral, longitudinal and vertical linear stiffness and damping primary and secondary suspensions. This investigated model governs lateral displacement, vertical displacement, roll yaw angles of each of wheelset and the lateral displacement, vertical displacement, roll and yaw angle of conventional truck. Kalker's linear theory has been adopted to evaluate the creep forces which are introduced on rail wheels due to rail wheel contact. The railway truck mathematical equations of motion are solved using fourth order Rung-Kutta method which requires that differential equations to be transformed into a set of first order differential equations. The transformed state space equations are simulated with computer aided simulation to represent the dynamic behavior and time solution of dynamics of conventional truck moving on tangent tracks. Influences of the geometric parameters of the rail wheel such as wheel conicity and nominal rolling radius on the dynamic stability of the system are investigated. It is concluded that the geometric parameters of the rail wheel have different effects on the hunting instability and on the change of the critical hunting velocity of the system. In addition critical hunting velocity of rail trucks is proportional inversely with the square roots of wheel conicity but high critical hunting velocity obtained by increasing the nominal rolling radius of the rail wheel.

Keywords: Rail wheel, Conventional truck, Critical hunting velocity, Tangent tracks, Lateral response, Yaw response.

Nomenclatures

a	Half of track gage, m
b_2	Half distance between longitudinal secondary suspensions, m
C_{px}	Viscous damping constant of longitudinal primary suspension, N.s/m
C_{py}	Viscous damping constant of lateral primary suspension, N.s/m
C_{pz}	Viscous damping constant of vertical primary suspension, N.s/m
C_{rail}	Lateral rail damping coefficient, N.s/m
C_{sx}	Viscous damping constant of longitudinal secondary suspension, N.s/m
C_{sy}	Viscous damping constant of lateral secondary suspension, N.s/m
d_p	Half distances between primary longitudinal suspensions, m
F_{ciw}	Creep forces in rail-wheel contact (i = right, left), N
F_{niw}	Normal forces in rail-wheel contact (i = right, left), N
F_{raili}	Force produced by rail (flange contact force), (i = right, left), N
F_{syt}	Lateral truck suspension force, N
F_{sywi}	Lateral wheelset suspension force, (i = front, rear), N
F_{szwi}	Vertical wheelset suspension force, (i = front, rear), N
f_{11}	Lateral creep coefficient, N
f_{12}	Lateral/spin creep coefficient, N.m
f_{22}	Spin creep coefficient, N.m^2
f_{33}	Forward creep coefficient, N
h_T	Vertical distance from wheelset centre to the lateral secondary suspension, m
I_{tx}	Truck mass moment of inertia about X-axis, kg.m^2
I_{tz}	Truck mass moment of inertia about Z-axis, kg.m^2
I_{wx}	Wheelset mass moment of inertia about X-axis, kg.m^2
I_{wy}	Wheelset mass moment of inertia about Y-axis, kg.m^2
I_{wz}	Wheelset mass moment of inertia about Z-axis, kg.m^2
K_{px}	Spring stiffness of longitudinal primary suspension, N/m
K_{py}	Spring stiffness of lateral primary suspension, N/m
K_{pz}	Spring stiffness of vertical primary suspension, N/m
K_{rail}	Lateral rail stiffness, N/m
K_{sx}	Spring stiffness of longitudinal secondary, N/m
K_{sy}	Spring stiffness of lateral secondary, N/m
L_b	Half distance between two wheelsets of the truck, m
L_{cs}	Vertical distance between lateral secondary suspension and centre of the carbody, m
L_s	Horizontal distance between vertical secondary suspension and centre of the carbody, m
M_{sxwi}	Longitudinal wheelset suspension moment, (i = front, rear), N.m
M_{szt}	Vertical truck suspension moment, N.m
M_{szwi}	Vertical wheelset suspension moment, (i = front, rear), N.m
m_t	Mass of truck, kg
m_w	Mass of wheelset, kg
r_o	Centred rolling radius of wheel, m
V	Railway forward velocity, m/s
W	Axle load, N
Y_t	Truck lateral displacement, m
Y_w	Wheelset lateral displacement, m
Z_t	Truck vertical displacement, m

Z_w	Wheelset vertical displacement, m
Greek Symbols	
δ	Flange clearance between wheel and rail, m
λ	Wheel conicity
ϕ_t	Truck roll angle, rad.
ϕ_w	Wheelset roll angle, rad.
ψ_t	Truck yaw angle, rad.
ψ_w	Wheelset yaw angle, rad.

1. Introduction

Rail vehicles are considered as the most important transportation systems in the modern society due to its ability to move many persons and heavy loads fast, safely and with low impact on the environment. The important factor to the passengers to use appropriate rail vehicles is ride comfort in addition to safety. Railway transportation system safety requires that derailment or the wheel flange climb during rail vehicle running is never be allowed while ride comfort requires a self excited lateral railway oscillation or what called hunting phenomenon is eliminated.

The classical hunting oscillation is a swaying motion of rail vehicle caused by the forward speed of the vehicle and by wheel-rail interactive forces due to wheel-rail contact geometry and friction creep characteristics.

Hunting is instability appears at higher speeds as an oscillation in the wheelset and other vehicle components such as trucks and carbody. These higher rail vehicle speeds known as the critical velocities in which the rail vehicle starts to hunt or starts to be instable equilibrium. Below a certain critical speed, the motion is damped out and above the critical speed the motion can be violent, damaging track and wheels, and potentially causing derailment.

Wheelset is the basic component of the rail vehicle responsible for safe and comfortable transportation and it is the most component in rail vehicle playing an important role in this undesirable derailment and hunting phenomenon which caused due to produced forces between the wheel and tracks. Most of these forces which are called the creep forces introduced according to friction properties of the wheel rail contact geometry. It is well known that the undesirable hunting instabilities of wheelset transform to the trucks and car body in which can be eliminated by increasing the critical velocities of the railway vehicle.

There are two types of trucks in the rail vehicle conventional and unconventional trucks. In which the rail wheels are attached together with a solid axle in conventional trucks so the rail wheels are dependently rotating and moving while the wheels are independently rotating and moving in the unconventional trucks. The important parameters in the rail wheel geometry are the wheel conicity and nominal rolling radius. Influences of these wheel parameters of geometry on the stability of the trucks are investigated in this research.

Many researches have been presented to study the dynamic behaviour and stability of rail vehicle components due to wheel rail contact creep forces at the critical hunting velocity while the parameters of the rail wheel geometry have taken into consideration. An early study on this subject was presented by Wickens [1, 2]

in which the dynamic stability of railway vehicle wheelsets and bogies having profiled wheels is investigated. It was shown that the dynamic instability of railway vehicle bogies and wheelsets is caused by the combined action of the conicity of the wheels and the creep forces acting between the wheels and the rails.

In the study presented by Suda et al. [3] the hunting stability and curving performance of high speed rail vehicles is represented considering non-linear creep force and flange contact. In addition, the lateral force of front wheel on outer rail and critical speed of wheelset are calculated and the influence of linkages between wheelset and truck frame were examined.

Conventional and unconventional wheelset were used into the research investigated by Jawahar et al. [4] in which nonlinear mathematical model used to evaluate the wheel-rail contact forces. In addition the results showed that unconventional system improves the dynamic features positively than the conventional system but at the same time the unconventional system shows a tendency for a constant lateral displacement of a wheelset from the centre.

Matsumoto et al. [5] reported that the measured creep characteristics in wheel-rail contact agree well with the calculated values based on Kalker's linear theory [6] and showed that a small change in wheelset lateral displacement may lead to a great change in creep characteristics.

The researches in which done by Ahmadian and Yang [7-9] investigated the Hopf bifurcation and hunting behaviour in a rail wheelset with flange contact and studied the effects of nonlinear longitudinal yaw damping on hunting critical speeds showing that large increasing in yaw damping can increase the hunting critical speeds and improve the hunting behaviour while Dukkipati and Swamy [10-11] considered modified railway passenger truck designs to improve the compatibility between the dynamic stability and the ability of the vehicle to steer around curves also the effects of suspension and wheel conicity were considered to evaluate the trade-off between dynamic stability and curving performance.

Nath et al. [12] studied the non-linear dynamics on railway wheelset moving on a tangent track and the governing equations of motion are derived using Lagrangian approach. It was shown in their research that the results presented are helpful to understand the complicated but important behaviour of wheelset and its dependence on axial velocity and yaw stiffness.

An important study is presented by Mohan [13] used controllable primary suspensions to improve hunting in railway vehicles moving on tangent tracks. In which single-point and two-point wheel-rail contact conditions were considered to study the dynamic responses of the rail vehicle components such as wheelset, trucks and car body. In addition the sensitivity of the critical speed to various primary suspension stiffness and damping parameters with different wheel conicities are examined and concluded that critical hunting velocity is proportional inversely to wheel conicity.

Using linear creep model Lee and Cheng [14] derived the governing differential equations of motion of a truck moving on tangent track to study the influences of suspension characteristics and wheel conicity to the critical hunting speed and compared the results with previous researches while in the study presented by Messouci [15] it was concluded that the lateral displacement and the yaw of both bogie frame and the wheelset are sensitive to variation of conicity

and their numerical values are less with a higher conicity and the reason is that the absolute value of the creep coefficients is higher**.** In addition the approach is based on providing guidance by creep forces in conjunction with wheel conicity, so that flange contact is normally avoided.

The main objective of the present study is to improve the dynamic behaviour of rail vehicle and eliminate the hunting instability of the trucks by increasing the critical hunting velocity. In which many parameters affect the stability of the trucks and change the critical hunting velocity such as primary and secondary suspension characteristics provided to the trucks and wheelsets, geometric properties of wheels and the tracks parameters. In the present study the influences of wheel geometry such as wheel conicity and nominal rolling radius are investigated and their effects on the critical hunting velocity of the trucks also represented.

The procedure achieved in this study is to construct rail truck model by deriving the concerning governing equations of motion and simulated with computer aided simulation. The constructed simulation model is used to study the influences of the wheel geometry such as wheel conicity and nominal rolling radius on the stability of the trucks.

2. Rail Vehicle Model

There are many rail vehicle models constructed according to the objective study in which are used but in this study the adopted rail vehicle model consists of single trucks with mass m_t and mass moment of inertia I_{tx}, I_{ty}, and I_{tz}, about X, Y, and Z axis respectively.

The truck is provided with two conventional single wheelsets each wheelset consists of two rigid steel wheels attached together to a solid axle in which the wheelset has mass m_w and mass moments of inertia I_{wx}, I_{wy}, and I_{wz}, about X, Y, and Z axes respectively. The truck system is equipped with two longitudinal primary suspensions of spring stiffness K_{px} and viscous damping constant C_{px} which are located distance $2d_p$ from each other. Also two primary lateral suspensions of spring stiffness K_{py} and viscous damping constant C_{py} and two vertical suspensions of spring stiffness K_{pz} and viscous damping constant C_{pz} which are located distance $2d_p$, are provided to the system.

Two secondary longitudinal suspensions are provided to connect the wheelset with the truck with spring stiffness K_{sx} and damping coefficient C_{sx} also there are two lateral secondary suspensions with spring stiffness K_{sy} and damping coefficient C_{sy}.

The truck model used in this study of 12 degrees of freedom and the associated governing equations of motion govern lateral displacement Y_t, yaw angle ψ_t, vertical displacement Z_t and roll angle ϕ_t of the truck while lateral displacement Y_w, yaw angle ψ_w, vertical displacement Z_w and roll angle ϕ_w of each wheelset as shown in Fig. 1.

The rail wheel geometry such as nominal rolling radius r_o and the wheel conicity λ are shown in Fig. 2.

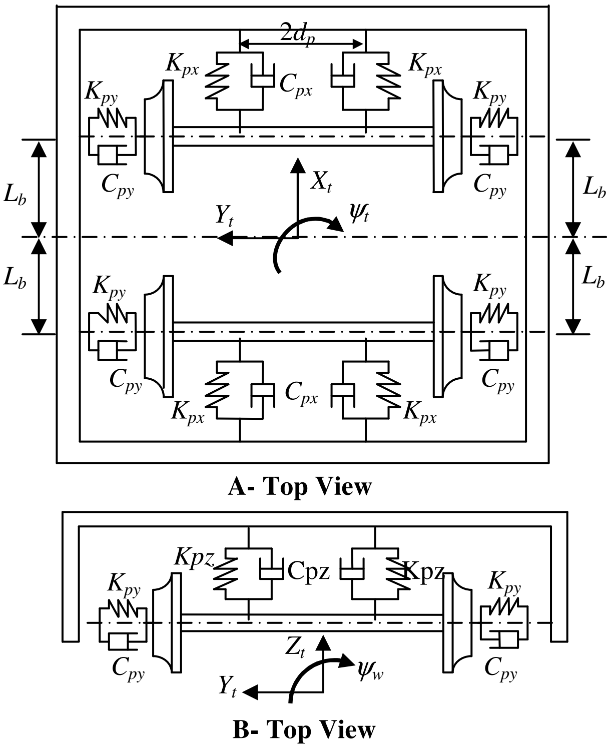

Fig. 1. Front and Top Views of Rail Conventional Truck Model with Longitudinal, Lateral and Vertical Suspensions.

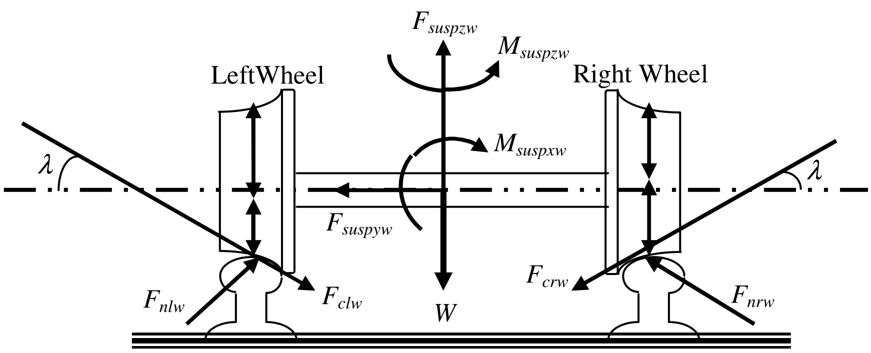

Fig. 2. Free Body Diagram of a Single Wheelst illustrates the Wheel Geometry such Nominal Rolling Radius r_o and the Wheel Conicity λ.

Differential equations of motion of rail truck model

There are several forces and moments affect the dynamic behaviour of the rail truck model most important of them are the creep forces which introduced on rail

and wheel contact patch area due to wheel rail geometry and due to friction characteristics between wheel and rail.

Linear Kalker's creep model [6] is used to evaluate the introduced creep forces and moments. In addition normal and suspension forces and moments also have to be evaluated in the governing equations. The lateral and yaw equations of motion of the truck are given by [14]

$$m_t \ddot{Y}_t = F_{syt} \tag{1}$$

$$I_{tz} \ddot{\psi}_t = M_{szt} \tag{2}$$

Roll and vertical differential equations of motion of the truck on tangent tracks are derived as follows:

$$m_t \ddot{Z}_t = -4C_{pz}\dot{Z}_t + 2C_{pz}\dot{Z}_{w1} + 2C_{pz}\dot{Z}_{w2} - 4K_{pz}Z_t + 2K_{pz}Z_{w1} + 2K_{pz}Z_{w2} \tag{3}$$

$$\begin{aligned} I_{tx}\ddot{\phi}_t &= -4C_{pz}L_s^2\dot{\phi}_t + 2C_{pz}L_s^2\dot{\phi}_{w1} + 2C_{pz}L_s^2\dot{\phi}_{w2} - 4C_{py}L_{cs}\dot{Y}_t + 2C_{py}L_{cs}\dot{Y}_{w1} \\ &+ 2C_{py}L_{cs}\dot{Y}_{w2} - 4K_{pz}L_s^2\phi_t + 2K_{pz}L_s^2\phi_{w1} + 2K_{pz}L_s^2\phi_{w2} - 4K_{py}L_{cs}Y_t \\ &+ 2K_{py}L_{cs}Y_{w1} + 2K_{py}L_{cs}Y_{w2} \end{aligned} \tag{4}$$

The lateral, yaw, vertical and roll equations of motion of the wheelset are given by [14]

$$m_t \ddot{Y}_{wi} = -\frac{2f_{11}}{V}Y_{wi} + 2f_{11}\psi_{wi} - \frac{2f_{12}}{V}\dot{\psi}_{wi} - W\phi_{wi} - \frac{2r_0 f_{11}}{V}\dot{\phi}_{w1} + F_{sywi} - F_{raili} \tag{5}$$

$$\begin{aligned} I_{wz}\ddot{\psi}_{wi} &= \frac{2a\lambda f_{33}}{r_0}Y_{wi} + \frac{2f_{12}}{V}\dot{Y}_{wi} + (-2f_{12} + a\lambda W)\psi_{wi} - \left(\frac{2a^2 f_{33}}{V} + \frac{2f_{22}}{V}\right)\dot{\psi}_{wi} \\ &+ \left(-\frac{I_{wy}V}{r_0} + \frac{2r_0 f_{12}}{V}\right)\dot{\phi}_{wi} + M_{szwi} \end{aligned} \tag{6}$$

$$m_w \ddot{Z}_{wi} = -\frac{2f_{11}}{V}\lambda^2\dot{\phi}_{wi}Y_{wi} - \frac{2f_{11}}{V}\phi_{wi}\dot{Y}_{wi} - \frac{2f_{12}}{V}\phi_{wi}\dot{\psi}_{wi} - \frac{2f_{11}r_o}{V}\dot{\phi}_{wi}\phi_{wi} + \frac{2f_{12}}{r_o}\lambda^2 + F_{szwi} \tag{7}$$

$$\begin{aligned} I_{wx}\ddot{\phi}_{wi} &= \left(\frac{2f_{12}\lambda^2}{r_0}\lambda^2 W\right)Y_{wi} - \frac{2f_{11}(r_0 + a\lambda)}{V}\dot{Y}_{wi} + \left(2f_{11}(r_0 + a\lambda) + \frac{2f_{22}\lambda^2}{r_0}\right)\psi_{wi} \\ &+ \left(\frac{I_{wy}V}{r_0} - \frac{2f_{12}r_0}{V} - \frac{2f_{12}a\lambda}{V}\right)\dot{\psi}_{wi} + (2\lambda^2 f_{12} + a\lambda W)\phi_{wi} \\ &- \left(\frac{2f_{11}ar_0\lambda}{V} + \frac{2f_{11}r_0^2}{V}\right)\dot{\phi}_{wi} + M_{sxwi} \end{aligned} \tag{8}$$

The suspension forces and moments are given by [14] as follows:

$$\begin{aligned} F_{syt} &= 2K_{py}Y_{w1} + 2C_{py}\dot{Y}_{w1} + 2K_{py}Y_{w2} + 2C_{py}\dot{Y}_{w2} \\ &+ (-4K_{py} - 2K_{sy})Y_t + (-4C_{py} - 2C_{sy})\dot{Y}_t \end{aligned} \tag{9}$$

$$\begin{aligned} M_{szt} &= (-4K_{py}L_b^2 - 4K_{px}d_p^2 - 2K_{sx}b_2^2)\psi_t + (-4C_{py}L_b^2 - 4C_{px}d_p^2 - 2C_{sx}b_2^2)\dot{\psi}_t \\ &+ 2K_{py}L_bY_{w1} + 2C_{py}L_b\dot{Y}_{w1} + 2K_{px}d_p^2\psi_{w1} + 2C_{px}d_p^2\dot{\psi}_{w1} - 2K_{py}L_bY_{w2} \\ &- 2C_{py}L_b\dot{Y}_{w2} + 2K_{px}d_p^2\psi_{w2} + 2C_{px}d_p^2\dot{\psi}_{w2} \end{aligned} \quad (10)$$

$$F_{sywi} = -2K_{py}Y_{wi} - (-1)^i 2K_{py}L_b\psi_t + 2K_{py}Y_t - C_{py}\dot{Y}_{wi} - (-1)^i 2C_{py}L_b\dot{\psi}_t + 2C_{py}\dot{Y}_t \quad (11)$$

$$F_{szwi} = -2K_{pz}Z_{wi} - 2C_{pz}\dot{Z}_{wi} \quad (12)$$

$$M_{sxwi} = -2K_{sy}h_TY_t - 2C_{sy}h_T\dot{Y}_t - 2d_p^2K_{pz}\phi_{wi} - 2d_p^2C_{pz}\dot{\phi}_{wi} \quad (13)$$

$$M_{szwi} = 2K_{px}d_p^2\psi_t - 2K_{px}d_p^2\psi_{wi} + 2C_{px}d_p^2\dot{\psi}_t - 2C_{px}d_p^2\dot{\psi}_{wi} \quad (14)$$

Force produced by the right and left rail which is called the flange contact force and is given in [16]:

$$F_{raili} = \begin{cases} K_{rail}(Y_{wi} - \delta) & Y_{wi} > \delta \\ 0 & -\delta \le Y_{wi} \le \delta \\ -K_{rail}(Y_{wi} + \delta) & Y_{wi} < -\delta \end{cases} \quad (15)$$

3. Numerical Simulation

Rail truck with two wheelsets moving on tangent tracks is modelled by the second order differential equations of motion; Eqs. (1) to (8). A simple and important technique used to transform the governing equations of motion from second order to first order differential equations in suitable form known as state space equations to facilitate solving them with numerical integration methods.

The transformed equations of motion are simulated with computer aided simulation to be solved by fourth order Runge-Kutta numerical method in which the general steps and the flow chart is illustrated in *Appendix A*.

Data used in numerical simulation is presented in *Appendix B* also the initial conditions are assumed for the dynamic motions of the system. Simulation is executed to study the dynamic responses of rail truck subjected to different parameters. Procedure is achieved by increasing the speeds until reaches the critical hunting velocity and using Lyapunov indirect method to study the stability of the system.

3.1. Rail-truck dynamic behavior

The dynamic behaviour of the rail truck due to different parameters of rail wheel geometry and suspension characteristics can be represented by the considered simulation model. Figure 3 shows the dynamic responses of the rail truck to lateral and vertical motions at speeds (155 km/h) below the critical hunting velocity with 0.05 wheel conicity. The lateral response of the rail truck is more sensitive dynamically than vertical response but vertical displacement has high amount.

Roll response of the rail truck as shown in Fig. 4 refers to high amplitudes of roll angles but vanished after a short time while yaw response has low amplitudes vanished after some long time and it is clear that roll motion has high amounts than the yaw motion of rail truck.

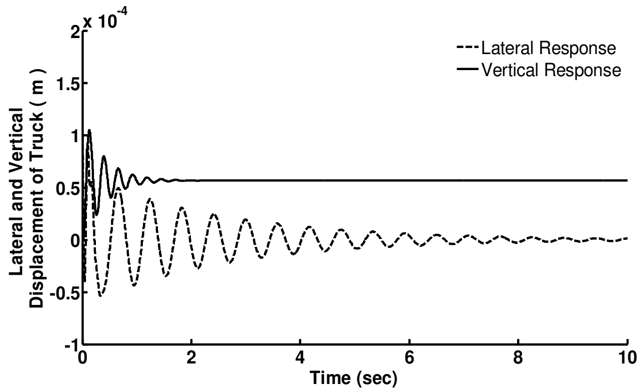

Fig. 3. Lateral and Vertical Displacement of Rail Truck at Speed (155 km/h) below the Critical Hunting Velocity with Wheel Conicity 0.05.

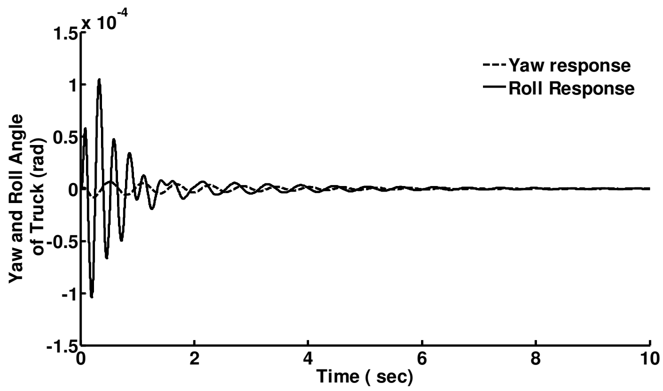

Fig. 4. Yaw and Roll Responses of Rail Truck at Speed (155 km/h) below the Critical Hunting Velocity with Wheel Conicity 0.05.

The dynamic responses of lateral and vertical motions of rail truck at critical hunting velocity (175 km/h) are depicted in Fig. 5 in which shows the lateral response is more sensitive to critical hunting velocity than lateral response.

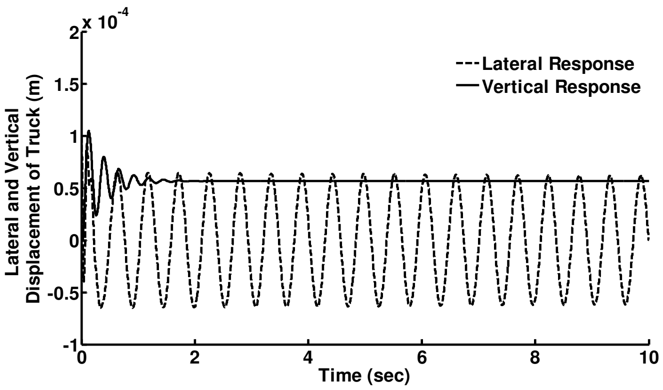

Fig. 5 Lateral and Vertical Responses of Rail Truck with Speed (175 km/h) at the Critical Hunting Velocity with Wheel Conicity 0.05.

Yaw and roll responses of the rail truck have different sensitive to critical hunting velocity as shown in Fig. 6 in which both motions have the same behavior to critical hunting velocity but more oscillations occur for short time in roll motion because longitudinal viscous damping coefficient C_{px} is more in magnitude than vertical viscous damping coefficient C_{pz} and that means damping through yaw motion is more rapid than roll motion.

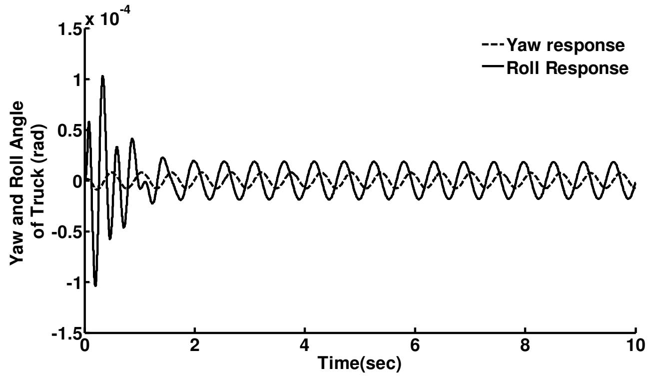

Fig. 6. Yaw and Roll Responses of Rail Truck with Speed (175 km/h) at the Critical Hunting Velocity with Wheel Conicity 0.05.

3.2. Influences of wheel geometry

The geometrical parameters of the wheel such as wheel conicity and nominal rolling radius affect the stability of rail truck and change the critical hunting velocity of the system.

Wheel conicity has a significant effect on the critical hunting velocity to eliminate the hunting phenomenon in railway truck. The effect of wheel conicity on critical velocity is shown in Fig. 7 in which represents that critical hunting velocity increases with low wheel conicity and as shown in Fig. 8 the critical hunting velocity of the system is proportional inversely to the square roots of the wheel conicity λ as presented also in [13].

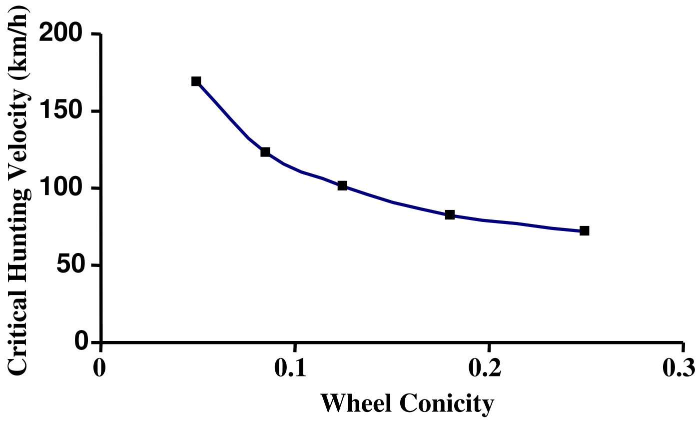

Fig. 7. Influence of Wheel Concity λ on Critical Hunting Velocity of Rail Truck Model moving on Tangent Tracks.

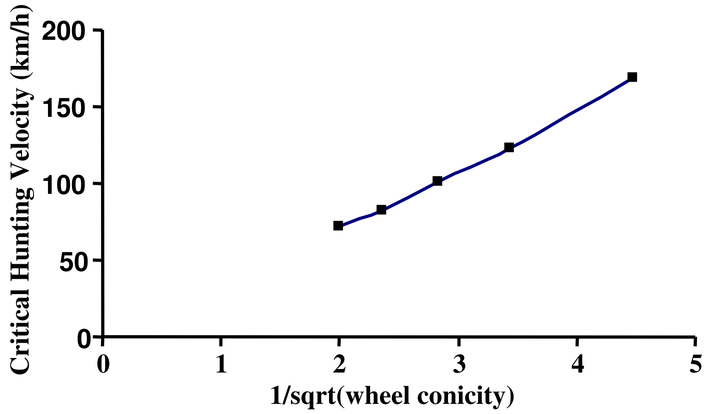

Fig. 8. Critical Hunting Velocity proportional inversely with Square Roots of Wheel Conicity of Rail Truck Model moving on Tangent Tracks.

The simulation model is used to represent the effect of the nominal rolling radius of wheels on the critical hunting velocity as shown in Fig. 9 in which high critical hunting velocity and better improvement of rail truck stability is obtained due to high nominal rolling radius of rail wheel.

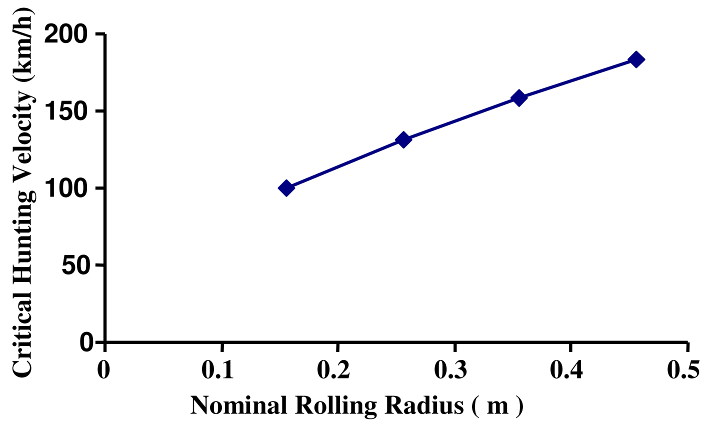

Fig. 9. Influence of Nominal Rolling Radius r_o on Critical Hunting Velocity of Rail Truck Model moving on Tangent Tracks.

4. Results and Discussion

The rail truck simulation model presents comparisons between the dynamic behaviour below and at critical hunting velocity of the truck responses subjected to different magnitudes of suspension parameters.

A compression between lateral and vertical dynamic response shows that lateral response is more sensitive to critical velocity than the vertical response since oscillations in lateral displacement take long time to be vanished to zero but

in vertical displacement the oscillations vanished within small time to a certain magnitude of vertical displacement. That means the simulation rail truck model gives acceptable magnitudes for the lateral and vertical displacements also the magnitudes of the suspension parameters used in this simulation model are limited and acceptable.

At the comparison study between roll and yaw displacement of the rail truck it can be noticed that oscillations in both displacements will be vanished to the stable point at zero displacement but with high amplitudes in roll response in which that depends upon the damping coefficient of the vertical suspensions.

The dynamic responses and oscillations of the rail truck model are increased when the system moves at the critical hunting velocity and it can be easily shown that the lateral displacement is the most sensitive dynamic response used to determine the critical hunting velocity of the system.

So in study of the influence of wheel geometrical parameters such as wheel conicity and nominal rolling radius the lateral dynamic response of the rail truck is used to determine the critical hunting velocity. It can be noticed that high critical hunting velocities with law magnitudes of wheel conicity but low critical hunting velocities with limited high nominal rolling radius.

5. Conclusions

Figures obtained from the simulation of rail truck model with 12 degrees of freedom show that:

- The model is able to represent the dynamic responses of rail truck to lateral, vertical, yaw and roll motion at different speeds of rail vehicle.
- Lateral and yaw response of rail truck are sensitive to change in rail vehicle forward speed and responsible for the critical hunting velocity and stability of rail truck more than vertical and roll responses. That refers to high influence of the introduced creep forces on the wheel due to wheel-rail geometry at lateral and yaw direction of motion than that creep forces introduced at vertical and roll direction of motion.
- Vertical and roll motions of rail truck has higher amounts than lateral and yaw motions in which can be interpreted that there are two groups of primary and secondary lateral and longitudinal suspensions used to control the lateral and yaw movements. Meanwhile one group of vertical primary and secondary suspension used to control the vertical and roll movements.
- It is concluded that lateral and yaw are the significant indication of the hunting phenomenon of rail truck.
- It is stated that geometrical parameters of wheel have important effects on the rail truck stability but with different rates.
- Finally the critical hunting velocity of the rail truck is increased and the hunting stability improved at low wheel conicity whereas the critical hunting velocity is increased at high nominal rolling radius.

References

1. Wickens, A.H. (1965). The dynamic stability of railway vehicle wheelset and bogies having profiled wheels. *International Journal of Solids and Structures*, 1(3), 319-341.
2. Wickens, A.H. (1965). The dynamic stability of a simplified four-wheel railway vehicle having profiled wheels. *International Journal of Solids and Structures*, 1(4), 385-406.
3. Suda, Y.; Fujioka, T.; and Iguchi, M. (1986). Dynamic stability and curving performance of railway vehicle. *Bulletin of the JSME*, 29(256), 3538-3544.
4. Jawahar, P.M.; Gupta, K.N.; and Echempati Raghu (1990). Mathematical modelling for lateral dynamic simulation of a railway vehicle with conventional and unconventional wheelsets. *Mathematical and Computer Modelling*, 14, 989-994.
5. Akira Matsumoto; Yasuhiro Sato; Machi Nakata; Masuhisa Tanimoto; and Kang Qi. (1996). Wheel-rail contact mechanics at full scale on the stand. *Wear*, 191(1-2), 101-106.
6. Kalker, J.J. (1979). Survey of wheel-rail rolling contact theory. *Journal of Vehicle System Dynamics*, 8(4), 317-358.
7. Yang, S.; and Ahmadian, M. (1996). The Hopf bifurcation in a rail wheelset with nonlinear damping. *ASME J. Rail Transportation*, 12, 113-119.
8. Ahmadian, M.; and Yang, S. (1997). Effect of suspension nonlinearities on rail vehicle bifurcation and stability. *ASME J. Rail Transportation*, 13, 97-106.
9. Ahmadian, M.; and Yang, S. (1998). Hopf bifurcation and hunting behavior in a rail wheelset with flange Contact. *Nonlinear Dynamics*, 15, 15-30.
10. Dukkipati, Rao V.; and Swamy, S.N. (2001). Non-linear steady-state curving analysis of some unconventional rail trucks. *Mechanism and Machine Theory*, 36(4), 507-521.
11. Dukkipati, Rao V.; and Swamy, S.N. (2001). Lateral stability and steady state curving performance of unconventional rail truck. *Mechanism and Machine Theory*, 36(5), 577-587.
12. Nath, Y.; Athre, K.; and Jayadev, K. (2003). Non–linear dynamics of railway wheelset. *National Nonlinear Systems and Dynamics Conference*, *NCNSD*.
13. Mohan, A. (2003). *Nonlinear investigation of the use of controllable primary suspensions to improve hunting in railway vehicles*. M.Sc. Thesis, Virginia Polytechnic Institute and State University, Virginia.
14. Sen-Yung Lee; and Yung-Chang Cheng (2005). Hunting stability analysis of high-speed railway vehicle trucks on tangent tracks. *Journal of Sound and Vibration*, 282(3-5), 881-898.
15. Messouci, M. (2009). Steady state motion of rail vehicle with controlled creep forces on curved track. *ARPN Journal of Engineering and Applied Sciences*, 4(2), 10-18.
16. Ahmadian, M.; and Yang, S. (1998). Effect of system nonlinearities on locomotive bogie hunting stability, *Journal of Vehicle System Dynamics*, 29(6), 366-384.

Appendix A

Runge-Kutta Method of Order 4

In numerical analysis, the Runge–Kutta methods are an important family of implicit and explicit iterative methods for the approximation of solutions of ordinary differential equations. These techniques were developed around 1900 by the German mathematicians C. Runge and M.W. Kutta. We state the Runge-Kutta method of order 4 which is widely used algorithm. For the initial value problems set

$x_i = a + ih$ For $i = 0, 1, 2, \ldots, n$ where $h = \dfrac{b-a}{n}$.

Also, we set $y(a) = y_o$. For $k = 0, 1, 2, \ldots, n-1$, we define

$$y_{k+1} = y_k + \frac{k_1 + 2k_2 + 2k_3 + k_4}{6}, \text{ where}$$

$$k_1 = hf(x_k, y_k), \qquad k_2 = hf(x_k + \frac{h}{2}, \frac{k_1}{2}), \qquad k_3 = hf(x_k + \frac{h}{2}, y_k + \frac{k_2}{2}),$$

$$k_4 = hf(x_k + h, y_k + k_3),$$

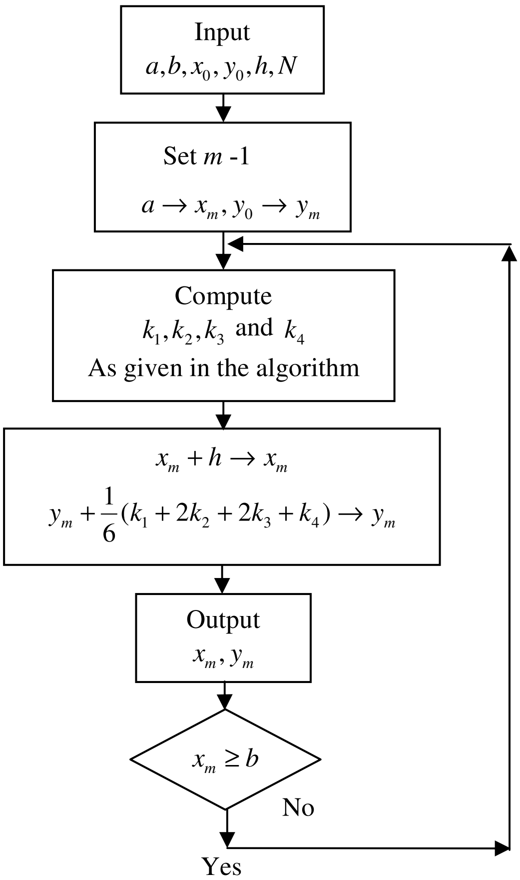

Fig. A-1. Flow-Chart of Runge-Kutta Method of Order 4.

Appendix B

Data used in Numerical Simulation

Table B-1. Railway Truck Model Data.

Symbol	Description	SI units	Value
a	Half of track gage	m	0.716
b_2	Half distance between longitudinal secondary suspensions	m	1.18
C_{px}	Viscous damping constant of longitudinal primary suspension	N.s/m	83760
C_{py}	Viscous damping constant of lateral primary suspension	N.s/m	45240
C_{pz}	Viscous damping constant of vertical primary suspension	N.s/m	3×10^4
C_{sx}	Viscous damping constant of longitudinal secondary suspension	N.s/m	9×10^4
C_{sy}	Viscous damping constant of lateral secondary suspension	N.s/m	1.8×10^3
d_p	Half distances between primary longitudinal suspensions	m	0.61
f_{11}	Lateral creep coefficient	N	9.43×10^6
f_{12}	Lateral/spin creep coefficient	N.m	1.2×10^3
f_{22}	Spin creep coefficient	N.m^2	10^3
f_{33}	Forward creep coefficient	N	10.23×10^6
h_T	Vertical distance from wheelset centre to the lateral secondary suspension	m	0.47
I_{tx}	Truck mass moment of inertia about *X*-axis	kg.m^2	3371
I_{tz}	Truck mass moment of inertia about *Z*-axis	kg.m^2	3371
I_{wx}	Wheelset mass moment of inertia about *X*-axis	kg.m^2	761
I_{wy}	Wheelset mass moment of inertia about *Y*-axis	kg.m^2	130
I_{wz}	Wheelset mass moment of inertia about *Z*-axis	kg.m^2	761
K_{px}	Spring stiffness of longitudinal primary suspension	N/m	2.85×10^5
K_{py}	Spring stiffness of lateral primary suspension	N/m	1.84×10^5
K_{pz}	Spring stiffness of vertical primary suspension	N/m	4.32×10^5
K_{rail}	Lateral rail stiffness	N/m	1.617×10^7
K_{sx}	Spring stiffness of longitudinal secondary	N/m	4.5×10^3
K_{sy}	Spring stiffness of lateral secondary	N/m	4.5×10^3
L_b	Half distance between two wheelsets of the truck	m	1.295
L_{cs}	Vertical distance between lateral secondary suspension and centre of the carbody	m	0.88
L_s	Horizontal distance between vertical secondary suspension and centre of the carbody	m	1
m_t	Mass of truck	kg	365
m_w	Mass of wheelset	kg	1751
r_o	Centred rolling radius of wheel	m	0.3556
V	Railway forward velocity	m/s	Variable
W	Axle load	N	5.6×10^6
δ	Flange clearance between wheel and Rail	m	9.23×10^{-3}
λ	Wheel conicity	-	0.05